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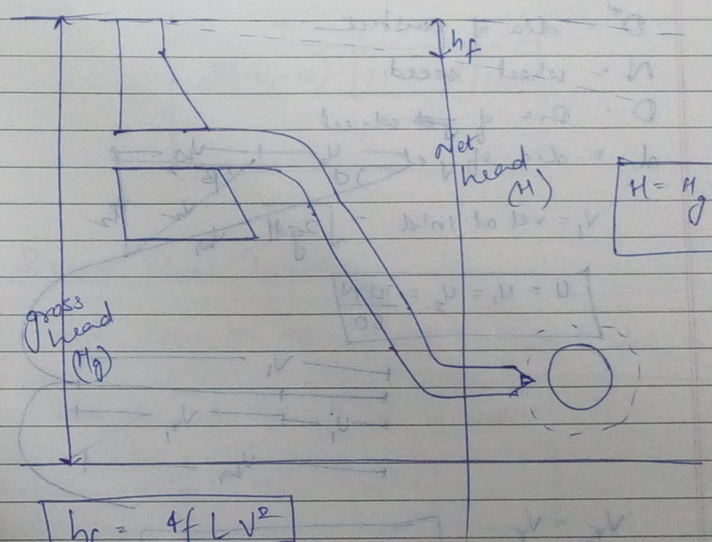
SR. NO.	TITLE	DATE	TEACHER'S SIGN / REMARK

Turbines



$$\boxed{H = H_g - h_f}$$

$$\boxed{h_f = \frac{4fLV^2}{2gD}}$$

Hydraulic Efficiency $\boxed{\eta_h = \frac{RP}{WP}}$ $\boxed{WP = \frac{W \times H}{1000}}$

$$\boxed{WP = \frac{\rho g Q H}{1000}}$$

Mechanical efficiency $(\eta_m) = \frac{SP}{RP} = \frac{\text{shaft power}}{\text{runner power}}$

Volumetric efficiency $= \frac{\text{vol. of water actually striking the runner}}{\text{vol. of water supplied to the turbine}}$

Overall efficiency $= \boxed{\eta_o = \frac{SP}{WP}} = \eta_h \times \eta_m$

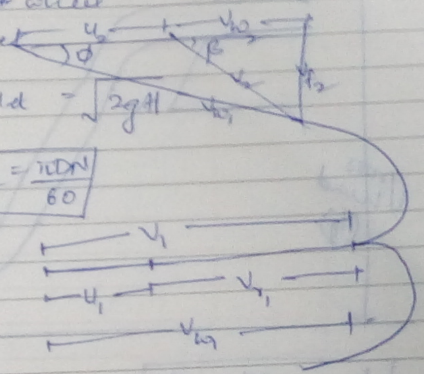
Pelton

D^* = dia of penstock

N = wheel speed

D = Dia of ~~jet~~ wheel

d = dia of jet



V_1 = vel at inlet $= \sqrt{2gH}$

$$\boxed{u = u_1 = u_2 = \frac{\pi D N}{60}}$$

$V_{r_2} = V_{r_1}$

$$\boxed{V_{w_2} = V_{r_2}\cos\phi - u_2}$$

$$F_x = \rho a V_1 \left[V_{w_1} + V_{w_2}\right]$$

work done per second $= \boxed{\rho a V_1 \left[V_{w_1} + V_{w_2}\right] u}$

power given to runner

work done/s per unit weight of water striking

$$= \boxed{\frac{\left(V_{w_1} + V_{w_2}\right) u}{g}}$$

$$\eta_h = \frac{2\left[V_{w_1} + V_{w_2}\right] u}{V_1^2}$$

$$= \boxed{\frac{2(V_1 - u)\left[1 + \cos\phi\right] u}{V_1^2}}$$

Radial Flow Turbines

$\boxed{\text{work done per sec} = \rho a V \left[V_{w_1}u_1 \pm V_{w_2}u_2\right]}$

work done p/s per unit wt. of water

$$= \frac{1}{g}\left[V_{w_1}u_1 \pm V_{w_2}u_2\right]$$

$$\boxed{\eta_h = \frac{V_{w_1}u_1 \pm V_{w_2}u_2}{gH}}$$

DR : ratio of pr. energy change in runner to total energy change

$\boxed{\Delta \text{ total energy} = \frac{1}{g}\left[V_{w_1}u_1 \pm V_{w_2}u_2\right]}$

$$\Delta = \boxed{\frac{V_1^2 - V_2^2}{2g} + \frac{u_1^2 - u_2^2}{2g} + \frac{V_{r_2}^2 - V_{r_1}^2}{2g}}$$

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$$R = \frac{(u_1^2 - u_2^2) + (V_{r_2}^2 - V_{r_1}^2)}{(V_1^2 - V_2^2) + (u_1^2 - u_2^2) + (V_{r_2}^2 - V_{r_1}^2)}$$

$$\boxed{R = 1 - \frac{(V_2^2 - V_1^2)}{2gH_e}}$$

For pelton wheel,

$u_1 = u_2 \qquad V_{r_2} = V_{r_1}$

$$\boxed{R = 0}$$

$$\boxed{R = 1 - \frac{\cot\alpha}{2(\cot\alpha - \cot\theta)}}$$

FLUID PROPERTIES

Fluid – substance that flows, doesn't have a shape of its own and takes the shape of container it is placed in.

$$\boxed{\text{Density} \quad \rho = \frac{\text{mass}}{\text{volume}}}$$

$$\boxed{\text{specific wt} = \rho \times g}$$

$$\boxed{\text{specific vol} = \frac{1}{\rho}}$$

$$\boxed{\text{specific gravity} = \frac{(\rho \times g) \text{ of reqd fluid}}{(\rho \times g) \text{ of standard fluid}}}$$

viscosity – resistance of fluid to motion

$$\mu \rightarrow \frac{\text{force} \times \text{time}}{\text{length}^2} \quad \left(\frac{Ns}{m^2}\right) \text{ or } (Pa \cdot s)$$

$$\boxed{1 \text{ poise} = \frac{\text{dyne} \times \text{sec}}{\text{cm}^2}}$$

$$\boxed{1 \frac{Ns}{m^2} = 10 \text{ poise}}$$

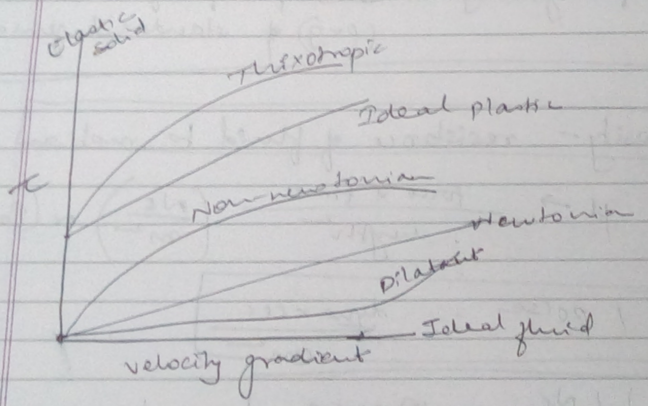
Kinematic viscosity $\boxed{\nu = \frac{\mu}{\rho}}$

unit $\frac{(\text{length})^2}{\text{time}}$

$\boxed{1\ \frac{m^2}{s} = 10^4 \text{ stoke}}$

Newtons law of viscosity
$$\tau = \mu \frac{du}{dy}$$

Liquids, μ reduces with temp.
Gases, μ increases with temp.



→ Newtonian $\tau = \mu \frac{du}{dy}$

→ Non-Newtonian

→ Purely viscous

→ Time-Independent

→ Pseudoplastic $\boxed{\tau = \mu \left(\frac{du}{dy}\right)^n}$, $n < 1$

→ Dilatant $\boxed{\tau = \mu \left(\frac{du}{dy}\right)^n}$, $n > 1$

→ Ideal Plastic/Bingham $\boxed{\tau = \tau_0 + \mu \left(\frac{du}{dy}\right)}$, $n=1$

→ Time-Dependent

→ Thixotropic $\tau = \mu \left(\frac{du}{dy}\right)^n + f(t)$

→ Viscoelastic

$\boxed{\tau = \mu \frac{du}{dy} + \alpha E}$

h = heat transfer coefficient
K = thermal conductivity.

CONDUCTION

Fourier's law of heat conduction —

$$\boxed{Q = -KA\frac{dT}{dx}}$$

General eqn for conduction of heat

$$\left[\frac{\partial}{\partial x}\left(K_x\frac{\partial t}{\partial x}\right)+\frac{\partial}{\partial y}\left(K_y\frac{\partial t}{\partial y}\right)+\frac{\partial}{\partial z}\left(K_z\frac{\partial t}{\partial z}\right)\right] \rightarrow \text{conduction heat}$$

$$+ \underbrace{[q_g]}_{\text{heat generated}} = \underbrace{\left[\rho c\frac{\partial t}{\partial \tau}\right]}_{\text{heat energy stored in element}}$$

if $K_x = K_y = K_z = K$,

$$\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}+\frac{\partial^2 t}{\partial z^2}+\frac{q_g}{K} = \frac{\rho c}{K}\frac{\partial t}{\partial \tau}$$

$$\frac{\rho c}{K} = \boxed{1/\alpha}$$

α = thermal diffusivity.

$$\boxed{\nabla^2 t + \frac{q_g}{K} = \frac{1}{\alpha}\frac{\partial t}{\partial \tau}}$$

Special cases –

① No heat generation

$$\nabla^2 t = \frac{1}{\alpha}\frac{\partial t}{\partial \tau}$$

② Steady state

$$\nabla^2 t + \frac{q_g}{K} = 0$$

further, in absence of internal heat generation,

$$\nabla^2 t = 0$$

Steady state, one dimensional, without internal heat generation

$$\frac{\partial^2 t}{\partial x^2} = 0$$

Cylindrical coordinates

$$\boxed{\left[\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{1}{r^2}\frac{\partial^2 t}{\partial \phi^2} + \frac{\partial^2 t}{\partial z^2}\right] + \frac{q_g}{K} = \frac{1}{\alpha}\frac{\partial t}{\partial \tau}}$$

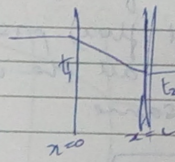
Spherical coordinates

$$\left[\frac{1}{r^2\sin^2\theta}\frac{\partial^2 t}{\partial \phi^2} + \frac{1}{r^2\sin\theta}\frac{\partial}{\partial \theta}\left(\sin\theta\frac{\partial t}{\partial \theta}\right) + \frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial t}{\partial r}\right)\right].$$

$$+ \frac{q_g}{K} = \frac{1}{\alpha}\frac{\partial t}{\partial \tau}$$

Heat conduction through a plane wall –

eqn → $\frac{d^2 t}{dx^2} = 0$ $\qquad \boxed{t = c_1 x + c_2}$



$$\boxed{t = \left(\frac{t_2 - t_1}{L}\right)x + t_1}$$

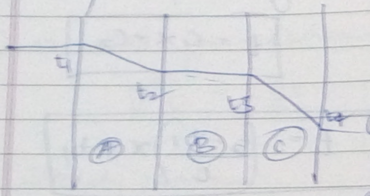
$$Q = \frac{t_1 - t_2}{\underbrace{(L/KA)}_{\text{(thermal resistance)}}}$$

Variable Thermal Conductivity :-

let $K = K_0(1+\beta t)$

$$Q = \underbrace{K_m}_{\text{mean (thermal conductivity)}} A\left[\frac{t_1 - t_2}{L}\right]$$

Composite Wall (Heat Conduction through) –



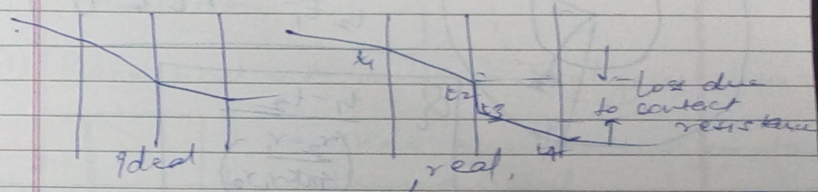
Constant factor, Q (heat flux) passing through all the walls will be same.

$$\boxed{R_A = \frac{L_A}{K_A A}} \quad \boxed{R_B = \frac{L_B}{K_B A}} \quad \boxed{R_C = \frac{L_C}{K_C A}}$$

$$\boxed{Q = \frac{(t_1 - t_4)}{R_A + R_B + R_C}}$$

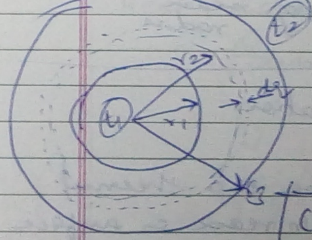
Thermal Contact Resistance –

Loss of temp. at interfaces, the so-called resistance that causes the same.



$$\boxed{\text{contact resistance} = \frac{t_2 - t_3}{(Q/A)}}$$

Conduction Through hollow cylinder –



$$\boxed{\frac{t - t_1}{t_2 - t_1} = \frac{\ln (r/r_1)}{\ln (r_2/r_1)}}$$

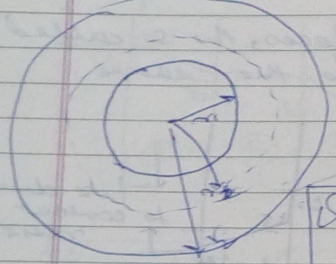
$$\boxed{Q = \frac{(t_1 - t_2)}{\left(\frac{\ln (r_2/r_1)}{2\pi K L}\right)}} \rightarrow R_{th} \text{ (thermal resistance)}$$

logarithmic mean area of cylinder –
A plane surface that would conduct the same amt. of heat as a given cylinder.

$$\boxed{A_m = \frac{A_o - A_i}{\ln (A_o - A_i)}}$$

(A_o, A_i – out/in areas of cylinder)

Heat conduction through sphere —



$$\boxed{\frac{t-t_1}{t_2-t_1} = \frac{r_2}{r}\left[\frac{r-r_1}{r_2-r_1}\right]}$$

$$\boxed{Q = \frac{t_1-t_2}{\left(\frac{r_2-r_1}{4\pi k r_1 r_2}\right)}}$$

Logarithmic mean area of hollow sphere -

$A_m = \sqrt{A_i A_o}$ $\qquad$ $r_m = \sqrt{r_1 r_2}$

r_m → log mean radius

Critical thickness of insulation

Insulation serves to increase the thermal resistance but if its too thick, the increase in surface area might serve to reduce the resistance. Thus a critical thickness exists where this transition occurs.

C.T for cylinder — $\boxed{r_c = \frac{k}{h_o}}$

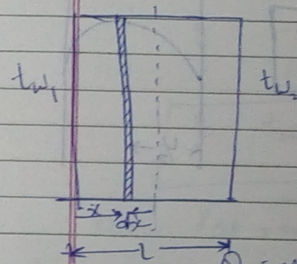
k → Thermal conductivity of insulating material

h_o → heat transfer coeff of outer surface of insulation

CT for sphere $= \frac{2k}{h_o}$

Heat Conduction with Internal Heat Generation —

① Plane wall with uniform heat generation —



There is uniform heat generation within the wall.

Q_x = heat conducted till x
$= -kA\frac{dt}{dx}$

Q_g = Heat generated in element $= A\,dx\,q_g$

Heat conducted till $x+dx$

$$Q_{x+dx} = Q_x + \frac{d}{dx}(Q_x)dx = Q_x + Q_g$$

$$\Rightarrow Q_g = \frac{d}{dx}(Q_x)dx$$

$$q_g A(\cancel{dx}) = \frac{d(Q_x)}{dx}(\cancel{dx})$$

$$q_g A = \frac{d}{dx}\left(-kA\frac{dt}{dx}\right) \Rightarrow \boxed{\frac{d^2t}{dx^2} + \frac{q_g}{k} = 0}$$

$$\boxed{t = -\frac{q_g}{k}x^2 + C_1 x + C_2}$$

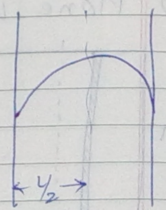
Case ① both sides have same temp.

at $x=0$ $t=t_w$

at $x=L$ $t=t_w$

$\Rightarrow \boxed{C_1 = \frac{q_g L}{2K}}$ & $\boxed{C_2 = t_w}$

$$\boxed{t = \frac{q_g}{2K}(L-x)x + t_w}$$



$$\boxed{t_{max} = \frac{q_g L^2}{8K} + t_w}$$

② case ② both sides have different temp

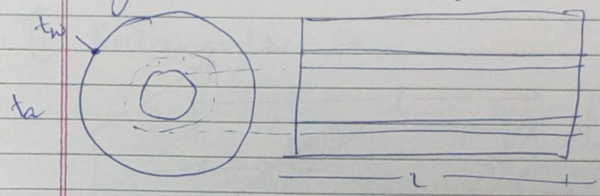
at $x=0$, $t=t_{w_1}$ $x=L$, $t=t_{w_2}$

$\boxed{C_2 = t_{w_1}}$, $\boxed{C_1 = \frac{t_{w_2}-t_{w_1}}{L} + \frac{q_g L}{2K}}$

$$\boxed{\frac{t-t_{w_2}}{t_{w_1}-t_{w_2}} = \left(\frac{q_g}{2K}\right)\left(\frac{L^2}{(t_{w_1}-t_{w_2})}\right)\left(\frac{x}{L}\right)\left(1-\frac{x}{L}\right)+\left(1-\frac{x}{L}\right)}$$

$$t = \left[\frac{q_g}{2K}(L-x) + \frac{t_{w_2}-t_{w_1}}{L}\right]x + t_{w_1}$$

② Cylinder with uniform heat generation



$$t = -\left(\frac{q_g}{K}\right)\left(\frac{r^2}{4}\right) + C_1 \log_e r + C_2$$

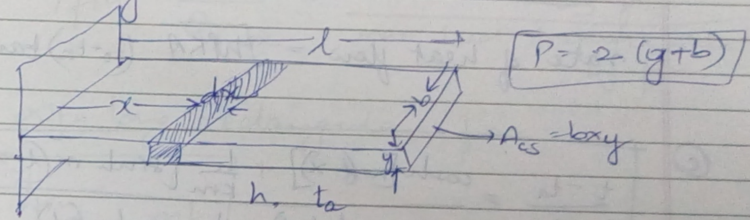
$$\boxed{t = t_w + \frac{q_g}{4K}[R^2 - r^2]}$$

$$t_{max}\ (\text{at } r=0) = t_a + \frac{q_g R}{2K} + \frac{q_g R^2}{4K}$$

$$m = \sqrt{\frac{hP}{KA}}$$

Heat Conduction through Fins –

1 Rectangular Fin



$$\theta = C_1 e^{mx} + C_2 e^{-mx} \qquad m = \sqrt{\frac{hP}{KA}}$$

Values of C_1 & C_2 depend on wether –

A fin is infinetly long

B end of fin is insulated.

C finite fin and loses heat by convection.

A
$$\boxed{\frac{t - t_a}{t_o - t_a} = e^{-mx}}$$

$$\boxed{Q_{fin} = \left(\sqrt{hPKA}\right)(t_o - t_a)}$$

[An infinite fin is the one for which $m \to \infty$ & this condition is approached when $ml > 5$]

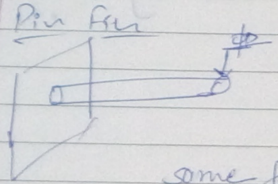
(b) $$\boxed{\frac{t-t_a}{t_0-t_a} = \frac{\cosh[m(l-x)]}{\cosh(ml)}}$$

rate of heat flow $= \sqrt{hPKA}\,(t_0-t_a)\tanh(ml)$

(c) $$\boxed{\frac{t-t_a}{t_0-t_a} = \frac{\cosh[m(l-x)] + \frac{h}{km}[\sinh m(l-x)]}{\cosh(ml) + \frac{h}{km}\sinh(ml)}}$$

$$\boxed{Q_{fin} = \sqrt{hPKA}\,(t_0-t_a)\left[\frac{\tanh(ml)+\frac{h}{km}}{1+\frac{h}{km}\tanh(ml)}\right]}$$

Pin fin



$$m = \sqrt{\frac{hP}{KA}} = \sqrt{\frac{4h}{KD}}$$

same formulae as for rectangular fins (all 3 cases).

Efficiency & Effectiveness of fins –

$$\eta_{fin} = \frac{Q_{fin}}{Q_{max}} = \frac{\text{actual heat transfered by fin}}{\text{max heat transferable by fin}}$$

(1) ∞ly long rectangular fin

$$\boxed{\eta_{fin} = \frac{1}{ml}}$$

(2) rect. fin with insulated tip

$$\boxed{\eta_{fin} = \frac{\tanh(ml)}{ml}}$$

where $ml = \sqrt{\frac{hP}{KA}}\,l = \sqrt{\frac{h(2w+2t)}{KA}}\,l$

if the fin is sufficiently wide $2w >> 2t$,

$\Rightarrow ml = \sqrt{\frac{2h}{Ktl}}\,l^{3/2} = \sqrt{\frac{2h}{KA_m}}\,l^{3/2}$

for rectangular fin, l is replaced by l_c

$l_c = l + t/2$.

→ Straight rectangular fin $(l_c = l + t/2)$ $(A_m = t l_c)$

→ Triangular fin $(l_c = l)$ $(A_m = l_c(t/2))$

→ Circumferential fin of rectangular cross section → $l = r_2 - r_1$

$l_c = l + t/2$

$r_{2c} = r_1 + l_c$

$A_m = (r_{2c} - r_1)\,t$

Effectiveness of fin

E = effectiveness = $\dfrac{Q_{\text{with fin}}}{Q_{\text{without fin}}}$

A_b = base area = area of surface on which fin was supposed to be there.

A_f = fin area.

$E = \dfrac{\tanh(ml)}{\sqrt{\dfrac{hA}{KP}}}$

if (ml) is sufficiently large, $\tanh(ml) \approx 1$,

$$\boxed{E = \sqrt{\frac{KP}{hA}}}$$

Effectiveness in terms of Biot No –

$$\boxed{E = \frac{1}{\sqrt{B_i}}\left[\frac{\sqrt{B_i} + \tanh[l\sqrt{B_i}]}{1 + \tanh[l\sqrt{B_i}]\sqrt{B_i}}\right]}$$

$B_i = 1 \Rightarrow E = 1$, fin doesn't matter

$B_i > 1 \Rightarrow E < 1$, fin acts as insulator, not desirable

$B_i < 1 \Rightarrow E > 1$, fin acts as conductor, desirable

Infinite fin $\eta = \dfrac{1}{ml}$

Finite (Insulated) =

Transient Conduction

t_i = initial surface temp.
t_a = ambient temp.

$$\boxed{\frac{t - t_a}{t_i - t_a} = e^{-\frac{hA\tau}{\rho V c}}}$$

h = unit surface conductance
A = surface area of body
τ = time
ρ = density
V = volume
c = specific heat of body

$\rightarrow$ $\frac{\rho V c}{hA}$ has dimensions of time

$\left(\frac{1}{hA}\right)(\rho V c) = R_{th} C_{th}$

$\frac{1}{hA}$ → Resistance to ~~[illegible]~~ convection heat transfer

$\rho V c$ → lumped parameter capacitance

$$\rightarrow \boxed{\frac{hA\tau}{\rho V c} = \left(\frac{hV}{KA}\right)\left(\frac{A^2 K \tau}{\rho V^2 c}\right) = \left(\frac{hL_c}{K}\right)\left(\frac{\alpha\tau}{L_c^2}\right)}$$

where $\boxed{\alpha = \frac{K}{\rho c}}$ = thermal diffusivity of solid.

L_c = characteristic length = $\frac{\text{vol. of solid}}{\text{surface area of solid}}$

$\boxed{\frac{hL_c}{K}}$ = biot number

$\boxed{\frac{\alpha\tau}{L_c^2}}$ = fourier no

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Instantaneous heat flow and total heat transfer –

$$Q = -hA(t_i - t_a)\, e^{-B_i F_o}$$

Total or cumulative heat transfer

$$Q' = \rho V C (t_i - t_a)\left[e^{-B_i F_o} - 1\right]$$

Response –
time taken by a thermocouple to attain source temperature.

$$\frac{\rho V C}{hA} = \text{time constant} = \tau^*$$

τ^* is time taken for temp change to reach 36.8% of its final value in response to a step change in temp.

Sensitivity – time taken for temp. difference to reach 63.2% of its initial value.

$$\text{Total heat }(Q) = -\rho C V (T - T_a)$$
$$= -mC\Delta T$$

$$\frac{hL_c}{K} \times \frac{\tau\alpha}{L_c^2}$$

$$\frac{h}{K}\,\frac{\tau K A^2}{\rho C V^2}$$

$Bi \rightarrow \infty$

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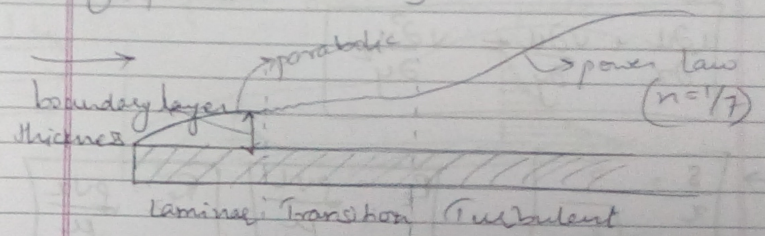
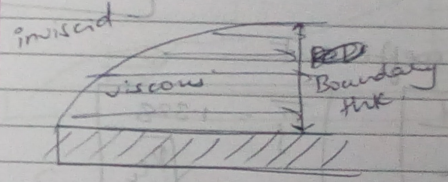
Boundary layer flow

U = free stream velocity





Boundary layer Thickness (δ)

= thk where velocity becomes 0.99U

→ Displacement thk (δ^*) = $\boxed{\int_0^{\delta}\left(1-\frac{u}{U}\right)dy}$

→ Momentum thk (θ) $\boxed{= \int_0^{\delta}\frac{u}{U}\left(1-\frac{u}{U}\right)dy}$

→ Energy thk (δ_e) = $\boxed{\int_0^{\delta}\frac{u}{U}\left(1-\frac{u^2}{U^2}\right)dy}$

Momentum Equation for Hydrodynamic Boundary layer Over A flat plate

$$\boxed{u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\frac{\partial^2 u}{\partial y^2}}$$

$$\boxed{\frac{\delta}{x} = \frac{5}{\sqrt{Re_x}}} \qquad \boxed{Re = \frac{\rho v L}{\mu}} \qquad \boxed{Re_x = \frac{\rho v x}{\mu}}$$

skin friction coefficient

$$\boxed{C_{f_x} = \frac{0.664}{\sqrt{Re_x}}} \qquad \boxed{\overline{C_f} = \frac{1.328}{\sqrt{Re_L}}}$$

at distance x — over entire plate of length L

$$\boxed{C_{f_x} = \frac{\tau_0}{\frac{1}{2}\rho U^2}}$$

$$\boxed{Re = \frac{\rho v L}{\mu}} \qquad \boxed{P_r = \frac{\mu C_p}{K}} \qquad \boxed{Nu = \frac{hL}{K}}$$

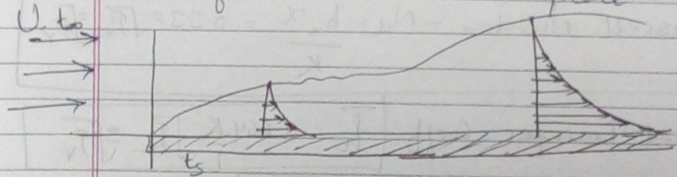
for flat plate

Laminar $Re < 5\times10^5$

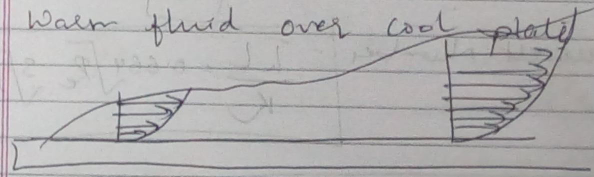
Turb. $Re > 5\times10^5$

Thermal Boundary layer

Cool fluid over warm plate



Warm fluid over cool plate



δ_{th} = thermal boundary layer, defined as the for which $\boxed{\frac{t_s - t}{t_s - t_\infty} = 0.99}$

Now, $P_r = \frac{\mu C_p}{K}$

$$\boxed{P_r = 1 \quad \delta_{th} = \delta} \qquad \boxed{P_r > 1 \quad \delta_{th} < \delta}$$

$$\boxed{P_r < 1 \quad \delta_{th} > \delta}$$

in general $\boxed{\delta_{th} = \frac{\delta}{\sqrt[3]{P_r}}}$

for laminar flow

Local Heat Transfer coeff $h_x = (0.332)\frac{K}{x}\sqrt{R_e}\sqrt[3]{P_r}$

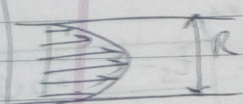
Local Nusselt Number $= N_u = \frac{h_x x}{K} = 0.332\sqrt{R_e}\sqrt[3]{P_r}$

Average Heat Transfer coeff $\bar{h} = 0.664\frac{K}{L}\sqrt{R_e}\sqrt[3]{P_r}$

$= 2h_x$

Average Nusselt Number $= \frac{\bar{h}L}{K} = 0.664\sqrt{R_e}\sqrt[3]{P_r}$

Laminar Tube Flow



$u_{max} = -\frac{R^2}{4\mu}\frac{\partial p}{\partial x}$

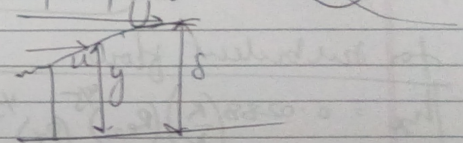
$\bar{u} = -\frac{R^2}{8\mu}\frac{\partial p}{\partial x}$

Head Loss over length L $= \frac{P_1 - P_2}{} = h_L = \frac{128\mu Q L}{\pi D^4}$

Turbulent flow

Forced convection over flat plate

$\frac{u}{U} = \left(\frac{y}{\delta}\right)^{1/7}$



$5\times10^5 < R_e < 10^7$

$\frac{\delta}{x} = \frac{0.371}{\sqrt[5]{R_{e_x}}}$

$\tau_0 = \frac{0.0288\rho U^2}{\sqrt[5]{R_{e_x}}}$

Local skin friction coeff $C_{f_x} = \frac{0.0576}{\sqrt[5]{R_{e_x}}}$

$\bar{C_f} = \frac{0.455}{(\log_{10} Re_L)^{2.58}}$

Total Drag due to laminar & Turbulent flow

$F_{total} = \bar{C_f} \times \frac{LB\rho U^2}{2}$

$\frac{0.455}{(\log_{10} Re_L)^{2.58}} - \frac{1670}{Re_L}$

Stanton No. $\boxed{St_x = \frac{Nu_x}{Re_x P_r}} = \boxed{\frac{C_{fx}}{2}}$

for turbulent flow

$$\boxed{h_x = 0.0288\left(\frac{K}{x}\right)(Re_x)^{4/5}(Pr)^{1/3}} \quad \boxed{\bar{h} = 0.036\left(\frac{K}{L}\right) Re^{4/5} P_r^{1/3}}$$

$$\boxed{Nu_x = 0.0288\, Re^{4/5} Pr^{1/3}} \quad \boxed{\overline{Nu} = 0.036(Re)^{4/5} P_r^{1/3}}$$

Radiation

Heat Transfer without intervening medium

Energy of photon $E = h\nu$

$h = 6.626 \times 10^{-34}$ J-S

① Total Emissive Power (E)

total amt. of radiation emitted per unit area & time

$E_b = \sigma T^4 = 5.67 \times 10^{-8} \frac{W}{m^2K^4}$

② Monochromatic Emissive Power (E_λ) —

rate of energy radiated per unit surface area per unit wavelength

$E = \int_0^\infty E_\lambda d\lambda \ \frac{W}{m^2}$

③ Emission from Real surface

$E = \underset{\downarrow}{\varepsilon} \sigma A T^4$

$\boxed{(\text{emissivity}) = \frac{\text{emissive power of a body}}{\text{emissive power of a blackbody}}}$

Total Radiation = $\underbrace{\alpha}_{\text{absorptivity}} + \underbrace{\rho}_{\text{reflectivity}} + \tau$ (transmittivity)

Stefan - Boltzmann law

$E_b = \sigma T^4$

Kirchoff's law

emissivity = absorptivity when a body is in thermal equb with surroundings.

$\alpha = \varepsilon = \frac{E}{E_b}$

Planck's law

$(E_\lambda)_b = \frac{2\pi c^2 h \lambda^{-5}}{e^{\left(\frac{ch}{\lambda k T}-1\right)}}$

Wien's Displacement law

$(E_b)_{max} = 1.285 \times 10^{-5} T^5 \frac{W}{m^2}$

$\lambda_m T = b = 2898$

Radiation exchange between 2 black bodies

$A_1 F_{1-2} = A_2 F_{2-1}$

Heat from 1 to 2 $= A_1 \sigma T_1^4$

Heat from 2 to 1 $= A_2 \sigma T_2^4$

Net heat exchange $= Q_{12} = A_1 F_{1-2} \sigma (T_1^4 - T_2^4)$

$\boxed{A_2 F_{2-1} \sigma (T_1^4 - T_2^4)}$

Radiation exchange b/w non-black bodies

→ 2 parallel planes

$\boxed{Q_1 = \frac{\varepsilon_1 \varepsilon_2}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1\varepsilon_2}}$ $\boxed{Q_2 = \frac{\varepsilon_2 \varepsilon_1}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1\varepsilon_2}}$

$Q_{1-2} = f_{12} \sigma (T_1^4 - T_2^4)$

interchange factor $= \boxed{\frac{1}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}}$

→ 2 parallel non-black cylinders

$\boxed{f_{12} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)}}$

$Q_{1-2} = f_{1-2} A_1 \sigma (T_1^4 - T_2^4)$

Electric Network Analogy for Thermal Radiation systems

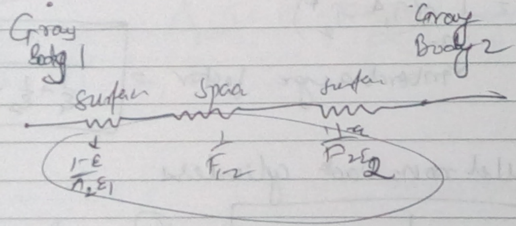
G = total incoming radiation
J = total outgoing radiation

$$\boxed{G = \frac{J - \varepsilon E_b}{1-\varepsilon}}$$

$$\boxed{Q_{net} = \frac{E_b - J}{\left(\frac{1-\varepsilon}{A\varepsilon}\right)}}$$ → surface resistance

$$\boxed{Q_{12} = \frac{J_1 - J_2}{\left(\frac{1}{AF_{1-2}}\right)}}$$ → space resistance

Total Resistance b/w 2 surfaces



$$(Q_{12})_{net} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{A\varepsilon_1} + \frac{1}{AF_{1-2}} + \frac{1-\varepsilon_2}{\varepsilon_2}\frac{A_1}{A_2}\frac{1}{A_1}}$$

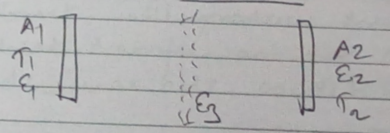
gray body factor

When radiating bodies are →

→ ∞ parallel planes $(A_1 = A_2)$ $F_{1-2} = 1$

→ concentric cylindrical sphere $F_{1-2} = 1$ $\frac{A_1}{A_2}$ = ratio of surface areas

Radiation Shield



Q_{12} = net radiatn. without shield $= \frac{A\sigma\left(T_1^4 - T_2^4\right)}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1\right)}$

$(Q_{12})_{net}$ = net radiation with shield of ε_3 $= \frac{A\sigma\left(T_1^4 - T_2^4\right)}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_3} - 1\right) + \left(\frac{1}{\varepsilon_3} + \frac{1}{\varepsilon_2} - 1\right)}$

If $\varepsilon_1 = \varepsilon_2 = \varepsilon_3 \Rightarrow$ $\boxed{T_3^4 = \frac{T_1^4 + T_2^4}{2}}$

resistance of n-shields put together

$$(R)_{n\text{-shield}} = \frac{(n+1)\left(\frac{2}{\varepsilon}-1\right)}{A}$$

$$(Q_{1-2})_{net} \text{ with n shields} = \frac{A\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}+2\sum_{i=1}^{n}\frac{1}{\varepsilon_i}-(n+1)}$$

Heat Exchangers

Components that are used to increase/decrease temp of working fluid using another secondary fluid.

Types

1. Parallel Flow
2. Counter flow

→ LMTD (logarithmic mean Temp difference)

It is that ~~d~~ temp. difference which, if kept constant, will produce the same amount of heat transfer as that actually occurs under variable conditions of ~~heat tra~~ temp. difference

Parallel Flow

$$\theta_m = \boxed{\frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)}}$$

Counter flow

$$\boxed{\theta_m = \frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)}}$$

Special case for counter flow $\theta_1 = \theta_2 = \theta$.

we take $\boxed{\theta_m = \theta}$.

U = overall heat transfer coefficient

$$Q = U A \theta_m$$

Q → total heat transfer

$$U = \frac{1}{\frac{1}{h_i} + \frac{L}{K} + \frac{1}{h_o}}$$

$$U_i = \frac{1}{\frac{1}{h_i} + \left(\frac{r_i}{K}\right)\ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_i}{r_o}\right)\frac{1}{h_o} + \left(\frac{r_i}{r_o}\right)R_{f_o}}$$

$$U_o = \frac{1}{\left(\frac{r_o}{r_i}\right)\frac{1}{h_i} + \left(\frac{r_o}{K}\right)\ln\left(\frac{r_o}{r_i}\right) + \frac{1}{h_o} + \left(\frac{r_o}{r_i}\right)R_{f_i}}$$

$$U_i A_i = U_o A_o$$

When fouling factor (R_f) is considered,

Question solving

$$Q = \dot{m}_{\text{hot fluid}} C_{p_h} (t_{h_1} - t_{h_2}) = \dot{m}_c C_{p_c} (t_{c_2} - t_{c_1}) = UA\theta_m$$

NTU is a measure of heat transfer size of heat exchanger. Larger the value of NTU, more the heat exchanger approaches its thermodynamic limit.

NTU (Number of Transfer Units) –

$$\text{Effectiveness of heat exchanger} = \frac{Q_{actual}}{Q_{maximum}}$$

Fluid capacity rate C

$$C_h = \dot{m}_h C_{p_h} \qquad C_c = \dot{m}_c C_{p_c}$$

$$C_{max} = \max(C_h, C_c) \qquad C_{min} = \min(C_h, C_c)$$

$$\varepsilon = \frac{C_c (t_{c_2} - t_{c_1})}{C_{min}(t_{h_1} - t_{c_1})}$$

$$Q = \varepsilon C_{min} (t_{h_1} - t_{c_1})$$

$$\frac{UA}{C_{min}} = NTU$$

$$R = \frac{C_{min}}{C_{max}}$$

Parallel Flow

$$\varepsilon = \frac{1 - e^{[-NTU(1+R)]}}{1+R}$$

Counter flow

$$\varepsilon = \frac{1 - e^{[-NTU(1-R)]}}{1 - R e^{[-NTU(1-R)]}}$$

Special Case

If $R = 1$

$$\varepsilon_{Parallel} = \frac{1 - e^{(-2NTU)}}{2}$$

$$\varepsilon_{Counter} = \frac{NTU}{1 + NTU}$$

If $R = 0$

$$\varepsilon_{both} = 1 - e^{(-NTU)}$$